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Active Control Of Surge In Compressors Which Exhibit Abrupt Stall

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ABSTRACT

The present paper reports a study on the dynamic behaviour of a compression system based on a multi-stage centrifugal blower and provided with a device for the active suppression of surge instability. The control device includes a sensor of total pressure at the compressor inlet, a proportional-derivative controller and an actuation valve at the plenum exit. A non-linear lumped-parameter model of the controlled compression system is presented, which considers also the time-lags in the compressor and actuator unsteady responses. The numerical simulation shows that: *i)* the predictions of control effectiveness provided by the non-linear model are rather different from the estimates of a linear approach, mainly due to the abrupt stall which affects the compressor characteristics; *ii)* the present control device is capable of suppressing surge within almost the whole unstable operating range of the compressor, with values of the proportional gain small enough to avoid actuator saturation (stroke-end occurrence); *iii)* the derivative component of the control law exerts a poor influence on the system stabilization and can be thus removed; *iv)* the proposed control strategy is effective over a wide range of stability parameter B , which implies the possibility of suppressing surge also at the highest compressor speeds and when using large plenum volumes. On the basis of the numerical results, a practical control device has been designed, which consists of a transducer of differential pressure between plenum and compressor exit (equivalent to a sensor of compressor face total pressure), a butterfly throttle/actuation valve driven by a stepper motor and a computer, provided with proper interfaces, for signal acquisition, data processing and motor control. The device has been installed in an industrial-size compression system and an exhaustive set of measurements has been planned in order to verify the effectiveness of the proposed control strategy and to validate the theoretical model.

INTRODUCTION

In the last decade much work has been devoted to the study of active suppression of compressor surge. Epstein et al. (1989) firstly proposed that surge can be prevented by actively damping the small disturbances which originate the instability while their amplitude is low. The active stabilization technique is based on the use of suitable sensor/actuator pairs in closed loop control devices. The output signal

from the sensor (of pressure, flow rate, etc.) is fed to a controller which applies a proper control law to drive the actuator (valve, moving plenum wall, etc.). The control is effective if the actuation is capable of absorbing and damping the unsteady energy surplus introduced in the system by the compressor when it operates in the stalled region. An experimental demonstration of active stabilization of surge has been given for centrifugal compressors by several investigators, for example, Ffowcs Williams and Huang (1989) and Pinsley et al. (1991). Furthermore, an extensive study has been carried out by Simon et al. (1993), who analyzed the influence of sensor and actuator selection on the maximum stabilized compressor characteristic slope by solving the linearized equations of the controlled system dynamics. According to this study, most of the literature on surge modelling and suppression employs a linear stability analysis to predict the effectiveness of active control devices. Such an approach provides reliable results as long as it can be assumed that surge instability originates from small amplitude perturbations, which is not the case of compressors which exhibit abrupt stall. This point has been examined closely by Giannattasio and Giusto (1998), with reference to a compression system provided with a device for the passive control of surge. They pointed out that the finite disturbance introduced by the sudden pressure drop associated with the abrupt stall causes non-linear effects to be responsible for system instability in conditions where the linear analysis predicts stable operation.

The present paper considers the active suppression of surge in a compression system based on a multi-stage centrifugal compressor which exhibits abrupt stall. This system has been employed for various experimental investigations in both stable and unstable operating conditions (Arnulfi et al., 1995, 1996). To study surge dynamics, a plenum of sufficiently large volume was connected to the compressor delivery pipe and detailed experimental data were collected on deep surge limit cycles and compressor internal flows (Arnulfi et al., 1999a). A non-linear lumped-parameter model of the compression system was also worked out, which proved to be capable of correctly predicting the system dynamics at different compressor speeds and throttle valve settings (Arnulfi et al., 1999b).

The active control device considered here includes a sensor of total pressure at the compressor inlet (or, equivalently, of differential pressure between plenum and compressor exit), a proportional-

derivative (PD) controller and an actuation valve at the plenum exit. The sensor/actuator pair has been selected on the basis of a theoretical analysis performed by Giannattasio (1999), who critically revised the work of Simon et al. (1993).

In the present study, a non-linear model and a preliminary experimental analysis of the actively controlled compression system are performed. The model is derived from the lumped-parameter approach used by Arnulfi et al. (1999b); the non-linear differential equations which describe the system dynamics are coupled to the control equations and are solved in the time domain by using an implicit second-order accurate numerical procedure. The dynamics of the controlled system has been simulated at various operating conditions of the compressor (flow rates and rotational speeds) and for different values of the control parameters (proportional and derivative gains). On the basis of the numerical results, a proper control device has been designed and installed in the experimental compression plant. At present, unsteady measurements of compressor and plenum pressure, mass flow rate and actuator position are being carried out in order to verify the control effectiveness in compressors which exhibit abrupt stall and to validate the proposed model of the system dynamics.

NOMENCLATURE

A	area
a	speed of sound
B	Greitzer parameter
E	surge unsteady energy
G	valve parameter
K_p	proportional gain
K_d	derivative gain
L	equivalent length
\dot{m}	mass flow rate
p	absolute pressure
t	time
T	period
T_H	Helmoltz period
U	impeller tip speed
u	system input = $(A_t - A_{ts})/A_{ts} = \Delta A_t/A_{ts}$
V	volume

Greek symbols

α_t	valve fraction open = $A_t/A_{t,max}$
φ	flow coefficient = $\dot{m}/\rho_0 U A_c$
ρ	density
τ	dimensionless time
τ_a	actuator time constant
τ_c	compressor time constant
ω_H	Helmoltz angular frequency
ψ	pressure coefficient = $2(p - p_0)/\rho_0 U^2$

Subscripts

0	ambient
1	compressor inlet
c	compressor

p	plenum
r	required
s	steady-state
t	throttle valve

SENSOR-ACTUATOR SELECTION

Figure 1 shows a sketch of a compression system fitted with the present device for the active suppression of surge. A signal of total pressure at the compressor inlet is fed to a PD controller, the output of which is employed to drive the throttle valve at the plenum exit. In unsteady conditions, the compressor face total pressure is directly related to the flow acceleration inside the compressor duct and can be thus considered as a representative output of the system during unstable operation. In the present case, the plenum valve performs both functions of throttling device and actuator. These functions can be kept apart by simply introducing a bleed valve, as the actuator, in parallel with the throttle valve (Simon et al., 1993). The two options are quite equivalent from a conceptual point of view, the best choice depending only on technical considerations. For example, it can be argued that the use of a bleed valve implies a gas leakage from the plenum also during steady operation. In fact, the actuator is required to move in both directions around the equilibrium position to make the control effective, so that the bleed valve cannot be completely closed in any operating condition.

The present sensor/actuator pair has been selected from the twelve different options suggested by Simon et al. (1993), who considered all the combinations of four sensors (compressor flow rate, plenum pressure, compressor face total and static pressure) and three actuators (close-coupled valve in the compressor delivery pipe, plenum bleed valve, movable plenum wall). On the basis of a linear stability analysis, Simon et al. attained to the conclusion that the combination of a sensor of compressor mass flow rate with a close-coupled valve is far the best choice for a control device of maximum effectiveness. However, such a result was obtained for fixed values of some operating parameters (steady equilibrium positions of throttle valve and close-coupled valve) which, moreover, were not reported in the paper. On the contrary, the variation of the steady operating point of an actual compression system was duly considered by Giannattasio (1999), who revised the comparative analysis of Simon et al. by using the same linear approach. The results of this study show that pair “compressor mass flow sensor/close-coupled valve” is clearly superior to the other ones only for small values of the close-coupled valve fraction open, which imply large pressure losses in the

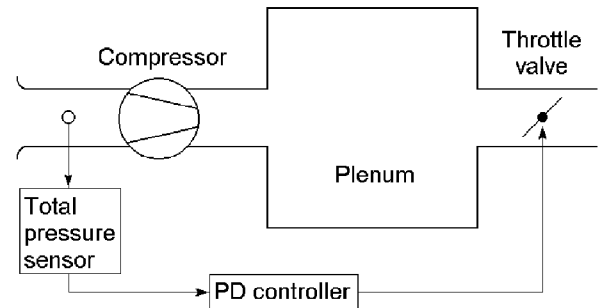


Fig. 1 - Controlled compression system.

compressor delivery pipe. Moreover, it turns out that five of the twelve sensor/actuator pairs have to be discarded because of severe gain-independent stability constraints, which cannot be removed by the control. Finally, some other combinations require extremely large values of the proportional gain for an effective system stabilization (compressor face total or static pressure/close-coupled valve) or involve technical complications (control of both speed and position of the movable plenum wall). In conclusion, the best compromise between control performance and technical requirements appears to be attained by the use of a sensor of total or static pressure at the compressor inlet and of an actuation valve at the plenum exit. The linear stability analysis shows that, when applied to the present compression system, the two pressure sensors produce equally effective controls. However, it can be shown that the compressor face total pressure is equal to the differential pressure between plenum and compressor exit, which is easier to be measured in unsteady conditions with the required accuracy. This explains the present sensor choice.

MODEL OF THE COMPRESSION SYSTEM

The dynamics of the controlled compression system is described by means of a non-linear lumped-parameter model derived from Greitzer (1976). This approach has been successfully adopted by the authors for modelling an uncontrolled compression system (Arnulfi et al., 1999b) as well as a compression plant provided with a novel device for the passive suppression of surge (Arnulfi et al., 2000).

The model is based on the following assumptions: *i*) one-dimensional incompressible flow in the compressor and valve ducts, which are replaced by constant area pipes of equivalent length L_c and L_t , respectively; *ii*) compressor regarded as an actuator disk; *iii*) uniform pressure and negligible flow velocity in the plenum; *iv*) quasi-steady flow through the valve. The further assumption, frequently retained, that the unsteady response of the compressor follows its steady-state characteristic is here replaced by the more realistic assumption of a time-lag which obeys to a first-order dynamics. The same approach is used to model the actuator dynamics, since the inherent bandwidth limitation of this component can introduce a non-negligible time lag between the controller output and the response of the flow train element.

The foregoing assumptions allow the following nondimensional equations of the system dynamics to be written (see, for details, Greitzer, 1976, Arnulfi et al., 1999b):

$$\frac{d\varphi_c}{d\tau} = B(\psi_c - \psi_p), \quad (1)$$

$$\frac{d\varphi_t}{d\tau} = \frac{B}{G}(\psi_p - \psi_t), \quad (2)$$

$$\frac{d\psi_p}{d\tau} = \frac{1}{B}(\varphi_c - \varphi_t), \quad (3)$$

$$\frac{d\psi_c}{d\tau} = \frac{1}{\tau_c}[\psi_{cs}(\varphi_c) - \psi_c], \quad (4)$$

$$\frac{du}{d\tau} = \frac{1}{\tau_a}(u_r - u), \quad (5)$$

$$\psi_t = (A_c/A_t)^2 \varphi_t^2 = (A_c/A_{ts})^2 \left(\frac{\varphi_t}{1+u} \right)^2, \quad (6)$$

$$u_r = \left(\frac{\Delta A_t}{A_{ts}} \right)_r = -K_p \psi_{01} - K_d \frac{d\psi_{01}}{d\tau}, \quad (7)$$

$$\psi_{01} = \psi_p - \psi_c. \quad (8)$$

Equations (1) and (2) are the momentum conservation equations in the compressor and valve pipes, respectively; Eq. (3) expresses mass conservation in the plenum; Eqs. (4) and (5) are first-order models of the compressor and actuator dynamics, respectively; Eq. (6) represents the steady-state characteristic of the throttle/actuation valve; Eq. (7) expresses the system input (nondimensional valve area) required by the PD control law; finally, Eq. (8) represents the system output (nondimensional total pressure at the compressor inlet) as a function of the system state variables (Simon et al., 1993).

Greitzer parameter B , which appears in Eqs (1)-(3), is defined as $B = U/2\omega_H L_c = (U/2a_p) \sqrt{V_p/A_c L_c}$ and can be interpreted as the ratio of pressure and inertial forces acting in the compressor pipe (Greitzer, 1976). The value of this parameter strongly affects the system stability and the control effectiveness. Valve parameter G in Eq. (2) is defined as $G = L_t A_c / L_c A_t$ and exerts a minor influence on the system dynamics (Greitzer, 1976). Term $\psi_{cs}(\varphi_c)$ in Eq. (4) refers to the steady-state compressor characteristic, while the time constant of the compressor dynamics, τ_c , can be related to the time needed for the complete development of a stall cell (Giannattasio and Giusto, 1998). Equation (5), which models the actuator dynamics, can be regarded as a first-order low-pass filter, its cut-off frequency being equal to the reciprocal of time constant τ_a (in the present work it is assumed $\omega_{cut} = 10\omega_H$ and hence $\tau_a = 0.1$). Finally, parameters K_p e K_d in control law (7) are real constants and represent the proportional and the derivative gains, respectively.

If ψ_t is eliminated from Eq. (2) by using Eq. (6) and Eqs (7)-(8) are introduced in Eq. (5), a non-linear system of five ordinary differential equations is obtained in the unknowns φ_c , φ_t , ψ_p , ψ_c and u . It can be solved numerically for given values of parameters B , G , τ_c , τ_a , K_p and K_d , if the steady-state characteristics of the compressor, $\psi_{cs}(\varphi_c)$, and of the throttle valve, A_{ts} , are known.

The present model is used to simulate the dynamic behaviour of an industrial compression system, which consists of a four stages centrifugal blower with vaned diffusers and a plenum of large capacity (detailed technical data on the present compression system are provided in paragraph "Experimental Procedure"). The steady-state characteristics of the compressor at the rotational speeds of 2000, 3000 e 4000 rpm are shown in Fig. 2. These curves show the presence of abrupt stall at all the considered rotational speeds. As observed by Giannattasio and Giusto (1998), such an occurrence strongly affects the dynamic behaviour of the system with and without control devices. The values assumed by stability parameter B at the three rotational speeds (0.304, 0.457 and 0.608) turn out to be far higher than the critical value of the present compressor, $B_{crit} = 0.06$ (Greitzer, 1976, Giannattasio and Giusto, 1998), which results in a great system tendency to instability and accounts for the deep surge conditions observed by Arnulfi et al. (1999a) also at the lowest compressor speeds.

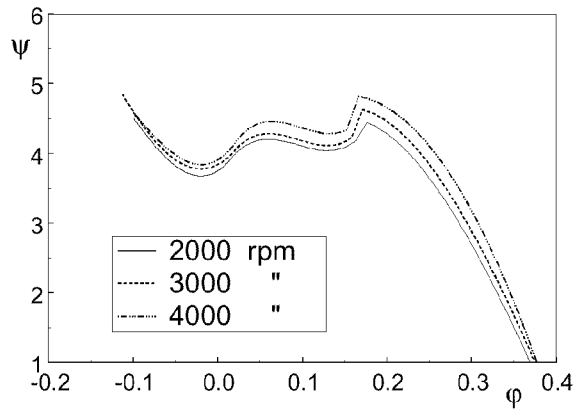


Fig. 2 - Compressor characteristic curves.

NUMERICAL RESULTS

The equations of the controlled system dynamics have been solved numerically by using an implicit second-order accurate time discretization and a Newton-Raphson procedure for the iterative solution of the resulting non-linear algebraic equations. The use of an implicit scheme is due to the stiffness of the present differential equation system, which can cause a severe numerical instability if an explicit discretization is employed.

The initial conditions are assigned by simulating a valve throttling process, with constant closing rate, from a steady operating point on the stable branch of the compressor characteristic. Differently from the assignment of an arbitrary perturbation of the steady equilibrium point, this procedure allows the effects of realistic initial conditions on the non-linear system dynamics to be taken into account.

At the end of each Newton-Raphson iteration, the possible occurrence of actuator stroke-ends is checked by verifying that the system input required by the controller (u_r in Eq. (7)) is within its limit values, corresponding to fully open or closed valve. If this is not the case, the control law is inhibited and the subsequent iterations are performed by fixing u_r at its limit value, until Eq. (7) yields again an achievable value of the system input. Furthermore, in the case of complete valve closure, Eq. (6) is no longer valid and it is replaced by equation $\phi_t = 0$.

Early numerical results were obtained by simulating single operating conditions of the compression system at different values of the proportional and derivative gains. As representative examples, Figs 3 through 5 show some results, obtained at the compressor speed of 3000 rpm, in terms of system trajectories in plane $\phi_c - \psi_p$ and time traces of variables ϕ_c , ψ_p and valve fraction open α_r . Part a) of each figure also shows, for reference, the steady-state characteristics of compressor (dashed line) and throttle valve (dot-dashed line).

Figure 3 refers to a steady operating point ($\phi_s = 0.15$) near the stall limit and to the case of purely proportional control ($K_d = 0$). This operating condition turns out to be normally unstable, since a deep surge limit cycle is observed when $K_p = 0$. However, it is noticed that a gain value of -10 is sufficient to stabilize the system, while higher absolute values of K_p result in an increasingly damped system response. As far as the actuator response is concerned, Fig. 3b shows

moderate oscillations of the valve fraction open around the steady equilibrium value, the actuator stroke-ends being never reached. In particular, the peaks of α_r represent the actuator response to the strong system perturbation caused by the crossing of the abrupt stall region.

Figures 4 and 5 refer to a steady operating point at a lower flow rate, namely, $\phi_s = 0.075$. Contrary to the prediction of the linear stability analysis (Giannattasio, 1999), this operating condition turns out to be unstable in the case of uncontrolled compression system, as shown in Fig. 4 by the deep surge cycle obtained at $K_p = 0$. Nevertheless, the inherent local stability of this point, promoted by the negative slope of the compressor characteristic, allows stability to be attained already at $K_p = -5$. Also in this case the oscillations of α_r have limited amplitude at all gain values and an overdamped actuator response is observed at the highest value of K_p . Figure 5, which represents the effects of the derivative gain on the system dynamics, shows that, as long as K_d assumes negative values, system stability is not affected. The main consequence of an increase in the derivative gain appears to be a smaller overshoot and an increased delay in the actuator response. On the contrary, a positive value of K_d can cause unstable operation, as in case c of Fig. 5a. This case shows a progressive departure of the system state from the steady equilibrium point, which finally results in small amplitude oscillations around a flow rate value much larger than the desired one.

In order to attain a global evaluation of the control device effectiveness, the unstable branch of the compressor characteristic ($0 < \phi_s < \phi_{stall}$) has been discretized in a sufficiently high number of equally spaced points, where stability has been tested at fixed values of K_p and K_d . By repeating this procedure for different values of the gains, contour-plots of the stabilized fraction of operating points in the gain plane have been obtained. The diagrams corresponding to the compressor speeds of 2000, 3000 and 4000 rpm are shown in Figs 6a, b and c, respectively, while Fig. 7 reports, for comparison, the analogous results obtained by the linear stability analysis (Giannattasio, 1999). Both linear and non-linear results show that negative values of K_p and K_d are required for system stabilization and that the derivative gain does not exert any significant influence on stability (the maximum extensions of the stable flow range can be obtained also with $K_d = 0$). Furthermore, it is observed that an increase in the compressor speed, and hence in parameter B , produces a rapid reduction in the control effectiveness, much larger absolute values of K_p being required to stabilize the same fraction of operating points. From a quantitative point of view, the predictions of the linear and non-linear models appear to be rather different. In fact, under uncontrolled conditions ($K_p = 0$), the contour-plots in Fig. 7 show that more than 40% of the considered flow range is stable, while the non-linear model predicts unstable operation everywhere. Nevertheless, as soon as the proportional control is introduced, the predictions of the non-linear model turn more optimistic than the linear ones. In fact, already at a comparatively small value of K_p ($K_p \approx -20$) both approaches provide the same estimate of the control effectiveness, while at higher absolute values of K_p , the stable flow range fraction predicted by the non-linear model turns out to be the larger. In any case, the proposed control strategy appears to be suitable to the stabilization of the present compression system, since 80-90% of the unstable branch of the compressor characteristic can be stabilized with reasonable values of the proportional gain.

The evaluation of the control effectiveness on the basis of the stabilized flow range fraction allows an immediate comparison between linear and non-linear approaches to be established, but it can

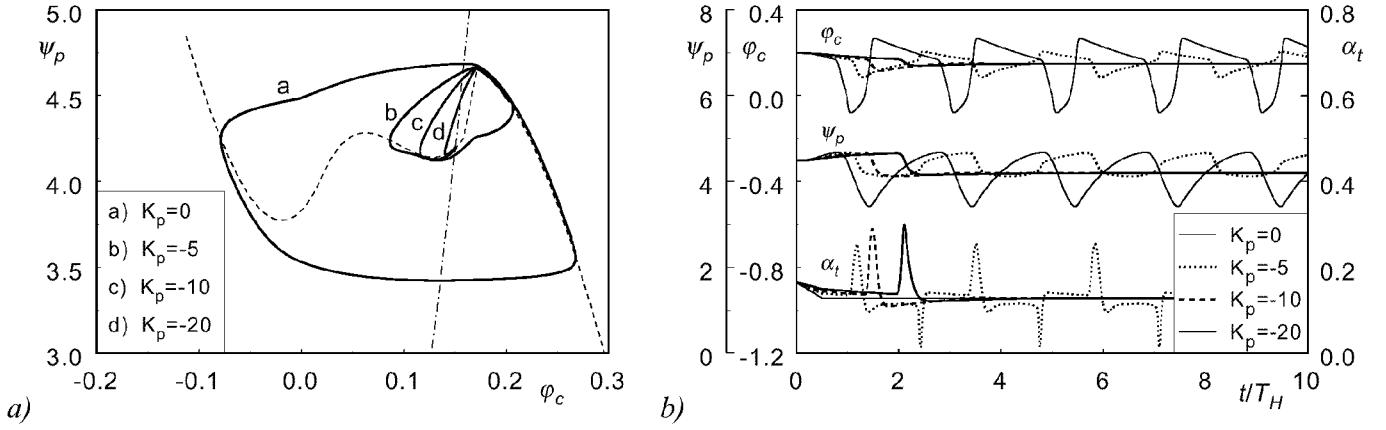


Fig. 3 - Simulated dynamics of the compression system at $n=3000$ rpm, $\phi_s=0.15$ and $K_d=0$ for different values of the proportional gain. a) System trajectories in ϕ_c - ψ_p plane. b) Time traces of compressor mass flow rate (ϕ_c), plenum pressure (ψ_p) and valve fraction open (α_t).

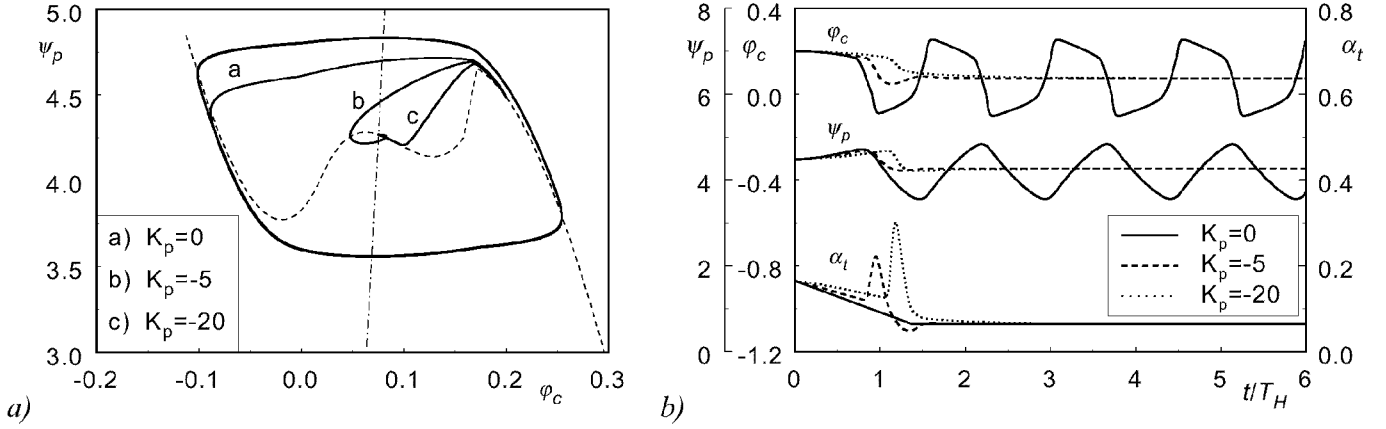


Fig. 4 - Simulated dynamics of the compression system at $n=3000$ rpm, $\phi_s=0.075$ and $K_d=0$ for different values of the proportional gain. a) System trajectories in ϕ_c - ψ_p plane. b) Time traces of compressor mass flow rate (ϕ_c), plenum pressure (ψ_p) and valve fraction open (α_t).

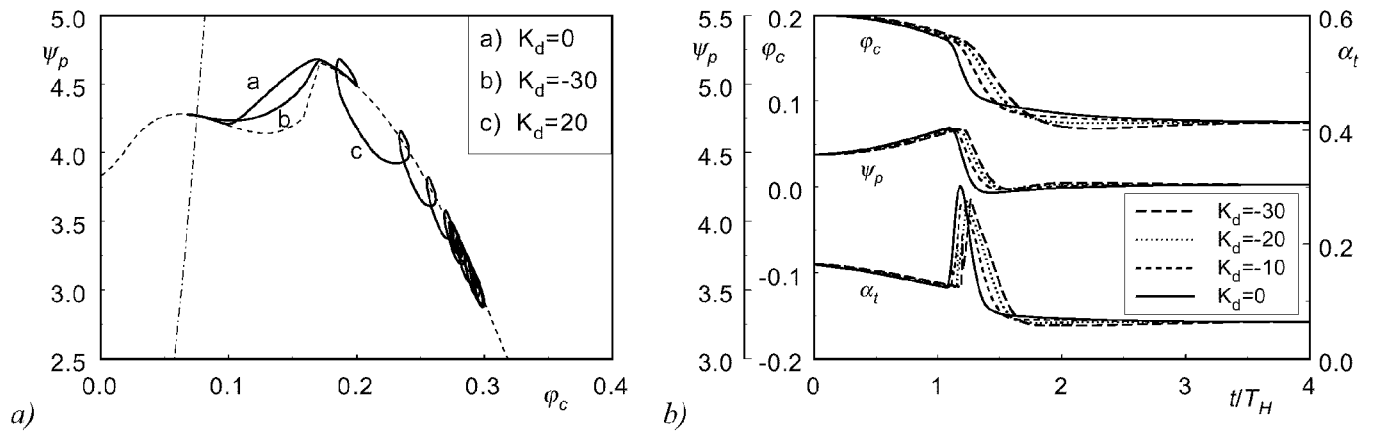


Fig. 5 - Simulated dynamics of the compression system at $n=3000$ rpm, $\phi_s=0.075$ and $K_p=-20$ for different values of the derivative gain. a) System trajectories in ϕ_c - ψ_p plane. b) Time traces of compressor mass flow rate (ϕ_c), plenum pressure (ψ_p) and valve fraction open (α_t).

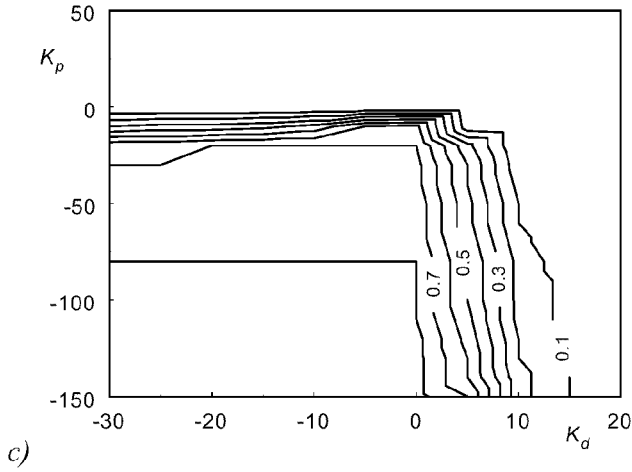
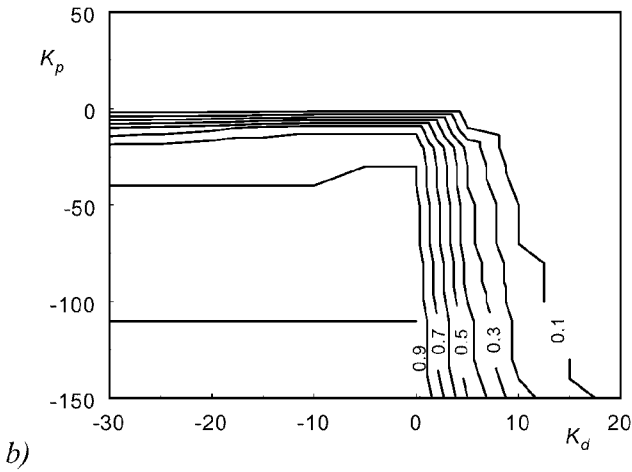
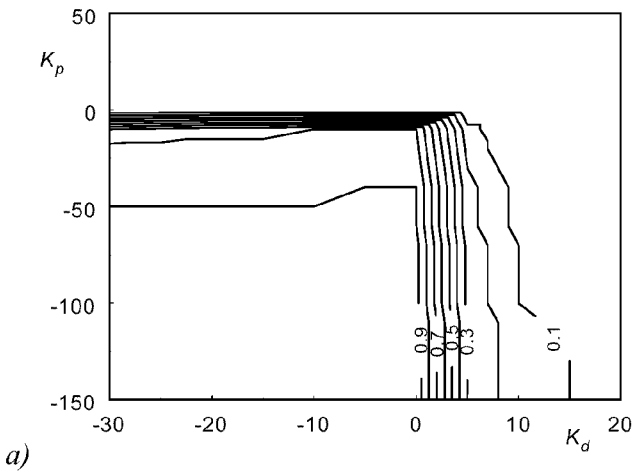


Fig. 6 - Contour-plots of the stabilized flow range fraction in the gain plane at the compressor speeds of a) 2000, b) 3000 and c) 4000 rpm. Results of the non-linear model.

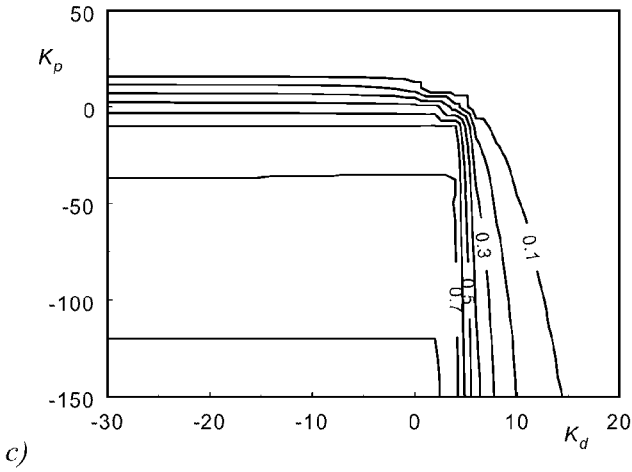
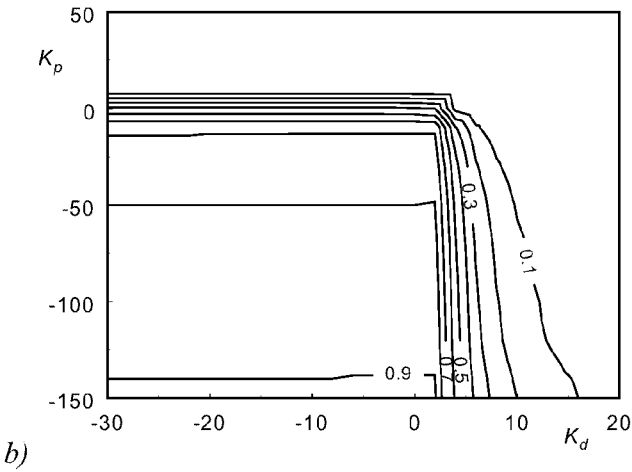
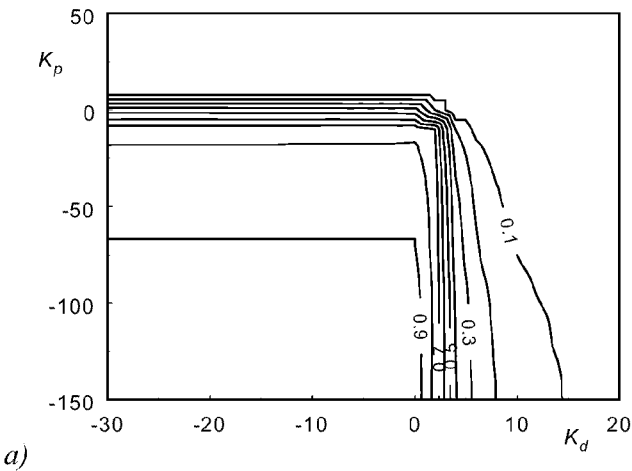


Fig. 7 - Contour-plots of the stabilized flow range fraction in the gain plane at the compressor speeds of a) 2000, b) 3000 and c) 4000 rpm. Results of the linear stability analysis (Giannattasio, 1999).

be inadequate to fully understanding the impact of the active control on the compression system dynamics. For example, the results in Fig. 6 provide no information about the changes in the amplitude of the surge cycles in the operating points which remain unstable after the active control has been introduced. In order to attain to a truly quantitative estimate of the control effectiveness, a proper stability index has been introduced by Giannattasio (1999), which is defined as:

$$E = \int_{\bar{\tau}}^{\bar{\tau}+T} (\psi_c - \psi_p) \frac{d\phi_c}{d\tau} d\tau \quad (9)$$

$$= B \int_{\bar{\tau}}^{\bar{\tau}+T} (\psi_c - \psi_p)^2 d\tau = \frac{1}{B} \int_{\bar{\tau}}^{\bar{\tau}+T} \left(\frac{d\phi_c}{d\tau} \right)^2 d\tau$$

where the integrals are computed over period T of a surge oscillation and the second and third equalities descend from Eq. (1). Parameter E can be interpreted as the unsteady energy associated with the surge cycle, being defined as the integral over the cycle period of perturbing “force” $(\psi_c - \psi_p)$ multiplied by “displacement” $d\phi_c$ (Greitzer, 1976). Furthermore, it can be immediately shown that, in the case of instantaneous compressor response ($\tau_c = 0 \Rightarrow \psi_c = \psi_{cs}$), E is equal to the area of the limit cycle in plane (ϕ_c, ψ_p) . It is clear that $E=0$ in a stable operating point, while high values of this parameter indicate an ineffective stabilization.

A useful representation of the control effectiveness can be obtained by plotting the values of E in all the operating points on the unstable branch of the compressor characteristic curve. This is done in Fig. 8, which refers to the compressor speed of 3000 rpm and shows E plots for different values of the proportional gain and $K_d=0$. According to the results in Fig. 6b, the uncontrolled system ($K_p=0$) turns out to be unstable all over the flow coefficient interval, E being different from zero everywhere. However, very small gain values are sufficient to stabilize a large part of the operating range, even if for $K_p > -20$ narrow unstable regions remain near the stall limit ($\phi_{stall} \approx 0.16$). At $K_p = -50$ the stabilized interval amounts to about 70% of the whole range, while further small increases require much larger absolute values of the gain. Finally, Fig. 8 shows that the active control produces a general reduction in the unsteady energy of the system, also in the operating points which remain unstable. As far as the actuator dynamics is concerned, it turned out that the valve does not reach its stroke-ends in any of the operating points considered. Consequently, the gain values in Fig. 8 can be assumed to be acceptable for a practical control device.

The present analysis has been generalized further on, by investigating the combined influences of parameters K_p and B on system stability. On the basis of the numerical results in Figs 5 and 6, the derivative gain was excluded from this parametric study as well as from the control law of the experimental device. In order to obtain a concise representation of the control effectiveness over the whole unstable flow range of the compressor, an average value of stability index E is here introduced, namely,

$$\langle E \rangle = \frac{1}{\phi_{stall}} \int_0^{\phi_{stall}} E d\phi. \quad (10)$$

Figure 9 shows a 3D diagram (a) and a contour-plot (b) of parameter $\log_{10}(1+\langle E \rangle)$ as a function of K_p and B (the logarithmic representation was preferred for a better graphic resolution at the

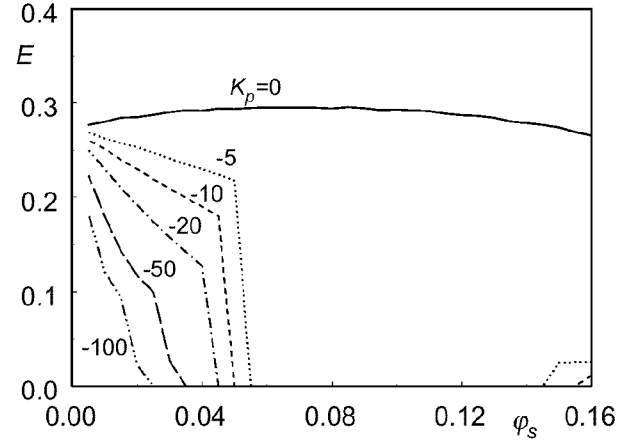


Fig. 8 - Unsteady energy E throughout the unstable flow range of the compressor characteristic at different values of the proportional gain ($K_d=0$, $n=3000$ rpm).

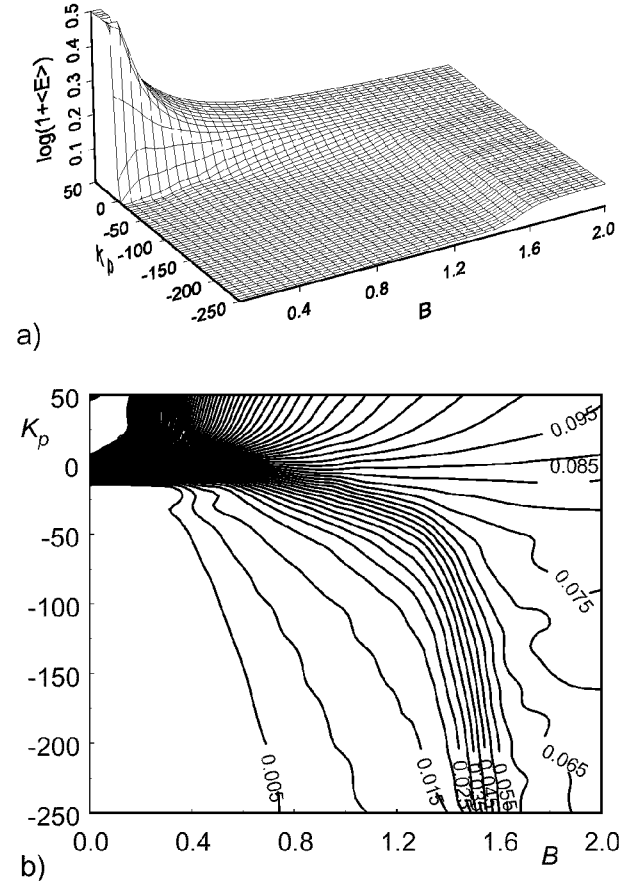


Fig. 9 - 3D plot (a) and contour-plot (b) of the average unsteady energy as a function of Greitzer parameter and proportional gain.

smallest values of $\langle E \rangle$). Both plots show that the region of low unsteady energy (high control effectiveness) lies in the range of negative gains and B values less than about 1.2. Inside this region stability index $\langle E \rangle$ increases very slowly with parameter B , which demonstrates the robustness of the proposed control strategy when applied to the present compression system. In fact, the maximum compressor speed is close to 4000 rpm, which corresponds to a B value of 0.608 for the present system geometry. This means that the stable operating range of the compressor can be enlarged considerably also under the heaviest conditions by using acceptable values of the proportional gain ($0 < K_p < 100$).

EXPERIMENTAL PROCEDURE

In order to verify the effectiveness of the proposed control strategy and to validate the theoretical model, an industrial compression plant has been coupled to a properly designed control device. A comprehensive set of measurements has been planned and preliminary tests are in progress.

The compression system, shown in Fig. 10, is installed at the Laboratory of the *Dipartimento di Energetica*, University of Trieste. It is based on a low pressure multi-stage centrifugal compressor driven by a DC motor through a speed increasing gear. The blower includes four impellers, with 16 backswept blades and 465 mm outer diameter, and vaned diffusers. The compressor inlet consists of a radial bellmouth duct with a 125 mm inner diameter ($A_c = 122.7 \text{ cm}^2$), while the delivery pipe is connected to a cylindrical plenum of large volume ($V_p = 3.132 \text{ m}^3$). The equivalent length of the compressor ducting turns out to be $L_c = 13.5 \text{ m}$. A butterfly valve at the plenum exit performs both functions of throttling device and actuator. The valve is 101 mm inner diameter and it is driven by a stepper motor with a resolution of 200 steps/rev. The motor is capable of delivering a torque higher than 11 Nm in the speed range of 100-500 steps/s.

The present compression system, without the control device, has been employed in a previous experimental work to perform detailed measurements of pressure, flow rate and flow velocity in deep surge conditions (Arnulfi et al., 1999a). Furthermore, the plant has been used to test the effectiveness of an innovative device for the passive control of surge (Arnulfi et al., 2000). This control device consisted of a hydraulic oscillator coupled to the plenum, the volume of which was reduced to 0.78 m^3 in order to obtain B values consistent with the stability limits of the device.

The instrumentation system is shown in Fig. 11, where the capital letters refer to the transducer locations in Fig. 10. Pressures and temperatures are measured by means of inductive transducers and K-thermocouples, respectively, while a magnetic pick-up is used for the compressor rotational speed. The mass flow rate is measured by means of an orifice flow meter mounted in the compressor delivery pipe. Although this instrument is normally used for steady flow measurements, it was considered acceptable in the present case because of the slow dynamics of surge (a few cycles per second).

The temperature signals are acquired by using a GPIB board as an interface to the HP3497A device. The pressure signals are first amplified and then conveyed to the NI SC-2040 board, which allows the simultaneous sampling of eight channels to be performed. This unit is an accessory of the multifunction I/O NI PCI-MIO-E1 board, which includes the A/D converter and is plugged in the PC (Pentium MMX 233MHz processor). The stepper motor is driven by a Power Driver Unit which allows a half-step resolution to be selected. The throttle angular position is measured by means of a precision

potentiometer with a linearity error of $\pm 0.25\%$. The software for data acquisition and motor control has been developed in LabView 5.0 environment.

The signal of differential pressure between plenum and compressor outlet is acquired at a sampling rate of 20 Hz, which was considered high enough to represent surge dynamics. The signal is first processed by means of a digital low-pass 2nd-order Bessel filter with a cut-off frequency of 8 Hz. A proportional control law is then applied to the filtered data by using selected values of K_p , so obtaining the valve flow area required by the control. This area value is used to compute the angular displacement of the actuator and hence the number of square-wave pulses to be sent to the stepper motor. The pulse frequency, which determines the motor speed, is set equal to 500 Hz, so allowing the required angular displacement of the actuator to be completed within a sampling period while enabling the motor to operate at high torque values. The signal from the potentiometer is used to provide an absolute reference for the angular position of the valve and to check the actual motor displacement.

The developed software allows the throttle valve to be moved to the desired position and the control device to be enabled or disabled when required, while recording the corresponding time evolution of the system parameters.

Preliminary experimental tests have been carried out in order to verify the operating features of the present control equipment. As a representative result, Fig. 12 shows the time traces of flow coefficient, plenum pressure coefficient, system output and valve angular position corresponding to a steady equilibrium point on the unstable branch of the compressor characteristic. The large amplitude oscillations at low frequency which are observed in the initial part of each diagram are due to surge. As soon as the control device is started, a significant reduction in the amplitude of the oscillations is obtained, but they are not completely suppressed. Perturbations of comparatively high frequency continue to affect the compression system, which cannot be considered as effectually stabilized. Further tests show that the amplitude of such oscillations becomes larger and larger as the absolute value of the gain is increased. Since higher gains are required to suppress surge at larger B values, the effects of the amplification of the high frequency disturbances rendered the system stabilization more and more ineffective as the compressor speed was increased beyond 2000 rpm.

The main reason for the lack of system stabilization appears to be the excessive time-lag (about 0.2 seconds) between the system output signal and the actuator response. Such a delay turns out to be comparable with the period of the high frequency perturbations observed under controlled conditions. Its main effect is an increased sensitivity of the system to measurement disturbances, which are amplified, rather than rejected, by high values of the gain.

The slow dynamics of the present control equipment is due both to the phase shift introduced by the Bessel filter and to software-hardware limitations which determine a low sampling frequency and a slow actuator response. Therefore, an effort has been undertaken aimed at improving the technical features of the present control system, by using more appropriate hardware and software tools and low phase-shift filtering techniques.

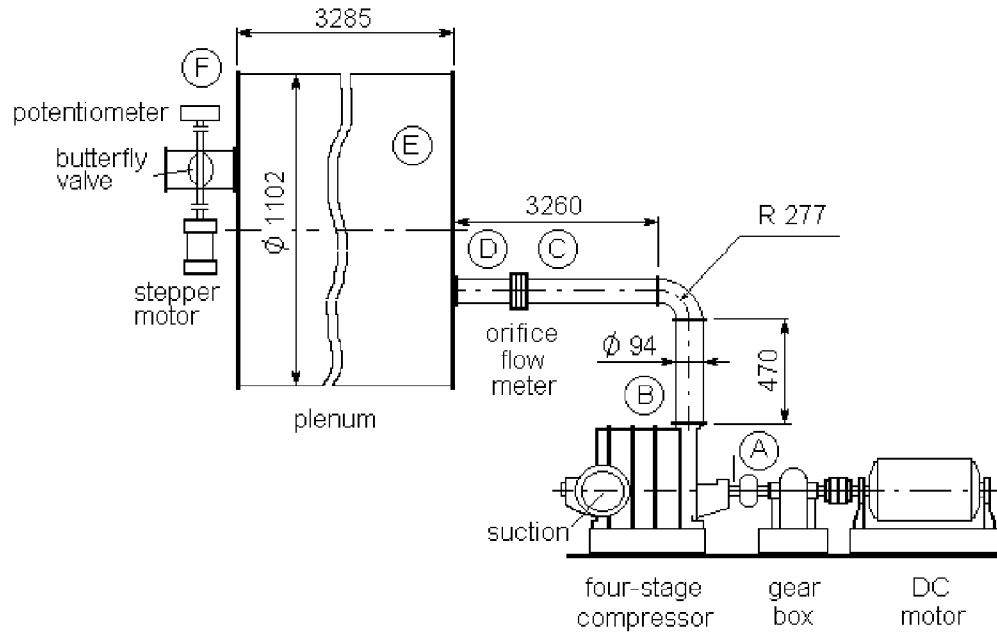


Fig. 10 - Experimental compression plant.

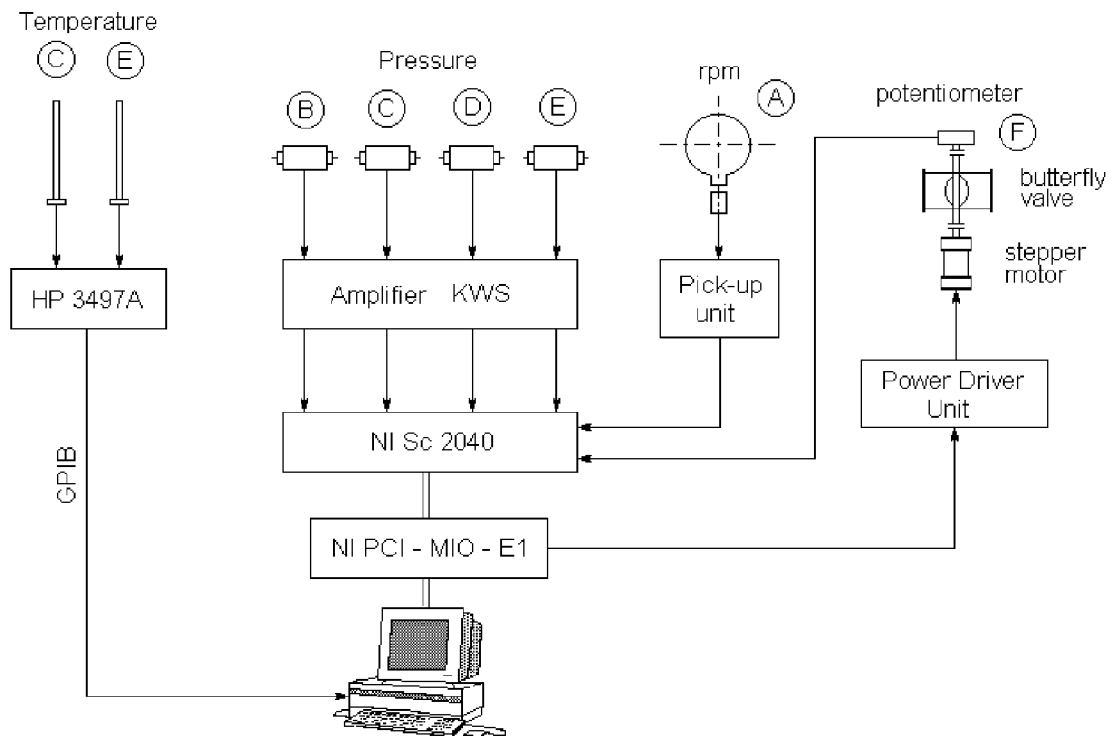


Fig. 11 - Schematic of the instrumentation system.

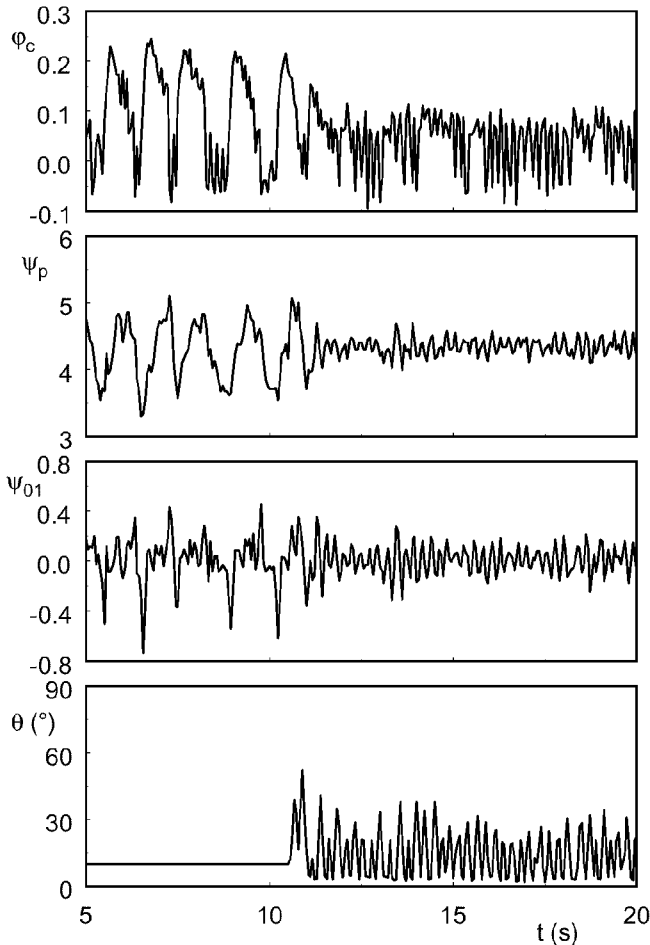


Fig. 12 - Preliminary experimental results at $n=2000 \text{ rpm}$, $\phi_s=0.03$ ($\theta_s=10^\circ$) and $K_p=-50$

CONCLUSIONS

The dynamic behaviour of an industrial compression system, provided with a proper control device for the active suppression of surge, has been investigated. The sensor/actuator pair has been selected on the basis of a linear stability analysis and technical considerations from previous works (Simon et al., 1993, Giannattasio, 1999) and results in a sensor of differential pressure between plenum and compressor outlet and a throttle/actuation valve at the plenum exit. A non-linear model of the controlled system has been developed which considers the general case of a proportional-derivative control law. The results of the numerical simulation allow the following conclusions to be drawn:

- A non-linear model must be used for a correct prediction of system dynamics and control effectiveness, at least in the case of compressors which exhibit abrupt stall. In fact, the finite amplitude perturbations introduced into the system when the stall limit is crossed strongly affect the system stability and cannot be predicted by a linear approach.

- The selected control device is capable of suppressing surge within almost the whole unstable operating range of the compressor, with values of the proportional gain small enough to avoid the actuator reaches its stroke-ends.
- The derivative component of the control law exerts a negligible influence on the system stabilization, while it produces only slight modifications in the transient evolution towards stability.
- The proposed control strategy is effective over a wide range of stability parameter B , which implies the possibility of suppressing surge also at the highest compressor speeds and when using plenum volumes also much larger than the present one.

On the basis of the numerical results, a proper control device has been designed and coupled to an industrial-size compression system. An appropriate instrumentation and data acquisition system has been set up and a comprehensive set of measurements has been planned.

Preliminary experimental results seem to confirm the effectiveness of the present control system, as the low-frequency surge oscillations are really suppressed by the device, at least at low compressor speeds and when using moderate gain values. However, an excessive delay in the actuator response is responsible for residual high frequency oscillations which are probably due to the amplification of measurement disturbances. The amplitude of such perturbations increases as the gain is raised, so preventing the compressor system from being stabilized at high compressor speeds. Therefore, further work is required to achieve an effective system stabilization within a wide operating range of the compressor, by improving the software efficiency and the technical features of the instrumentation system.

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